Assessment of the Influence of Design Parameters of a Pneumatic Drive on the Energy Efficiency of the Working Process

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Abstract. The aim of the study is to assess the impact of the pneumatic system structure and the choice of control algorithm for pneumatic distributors on the feasibility of implementing an energy-saving operating mode. This mode involves converting the kinetic energy of moving actuator masses into the potential energy of compressed air, with subsequent recuperation into the network. The objective is achieved by solving the following tasks: implementing pulse and relay control algorithms for various pneumatic drive configurations; investigating the influence of system parameters and control algorithms on the transient process shape and the ability to recuperate energy into the network during braking. During the course of these tasks, it was found that the shape of the transient process is influenced by the braking start moment and its duration, as well as the initial pressure level in the working (piston) chamber. The most significant result is the identification of pneumatic drive structural schemes and control algorithms that enable energy recuperation into the main line during braking, thereby providing a positive energy-saving effect. An analytical description of the sources of compressed air savings during drive operation is also provided. The significance of the results lies in the fact that the obtained schemes, which include braking by altering the connection structure of the pneumatic cylinder chambers, make it possible to implement an energy-saving operating mode. In this mode, the kinetic energy of moving actuator masses is converted into the potential energy of compressed air and is recuperated into the network. This enables further use of the recovered energy, enhancing the overall energy efficiency of the system.
***Keywords*:** pneumatic drive, control algorithm, switching, energy recovery, transient process.

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Evaluarea influenței parametrilor de proiectare a unui actionare pneumatice asupra eficienței energetice a procesului de lucru

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Rezumat. Scopul cercetării este evaluarea influenței structurii sistemului pneumatic și a alegerii algoritmului de control al distribuitoarelor pneumatice asupra posibilității de implementare a unui regim de funcționare economisitor de energie. Acest regim constă în transformarea energiei cinetice a maselor în mișcare ale acționarei în energie potențială a aerului comprimat, cu recuperarea ulterioară a acesteia în rețea. Scopul propus este atins prin rezolvarea următoarelor sarcini: implementarea algoritmilor de control impulsivi și în releu pentru diferite scheme ale acționărilor pneumatice; investigarea influenței parametrilor sistemului și a algoritmilor de control asupra formei procesului tranzitoriu și realizarea recuperării energiei în rețea în timpul frânării. În procesul de rezolvare a acestor sarcini s-a constatat că forma procesului tranzitoriu este influențată de momentul începerii frânării și de durata acesteia, precum și de nivelul inițial al presiunii în camera de lucru (piston). Cel mai important rezultat constă în determinarea schemelor structurale ale acționării pneumatice și a algoritmilor de control care permit realizarea regimului de recuperare a energiei în magistrală în perioada frânării, ceea ce oferă un efect pozitiv de economisire a energiei, precum și o descriere analitică a surselor de economie de aer comprimat în timpul funcționării acționării. Importanța rezultatelor obținute constă în faptul că schemele propuse, cu frânare realizată prin schimbarea structurii conexiunii camerelor cilindrului pneumatic, permit implementarea unui regim de funcționare economisitor de energie, în care energia cinetică a maselor în mișcare este transformată în energie potențială a aerului comprimat și este recuperată în rețea. Acest lucru permite utilizarea ulterioară a energiei recuperate, crescând nivelul de eficiență energetică a sistemului.
Cuvinte-cheie: acționare pneumatică, algoritm de control, comutare, recuperare de energie, proces tranzitoriu.

Оценка влияния конструктивных параметров пневматического привода на энергоэффективность рабочего процесса

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Аннотация. Целью исследования является оценка влияния структуры пневмосистемы и выбора алгоритма управления пневмораспределителями на возможность реализации энергосберегающего режима работы, заключающегося в преобразовании кинетической энергии движущихся масс привода в потенциальную энергию сжатого воздуха с последующей рекуперацией ее в сеть. Поставленная цель достигается за счёт решения следующих задач: реализация импульсных и релейных алгоритмов управления для различных схем пневмопривода; исследование влияния параметров системы и алгоритмов управления на форму переходного процесса и осуществление рекуперации энергии в сеть в процессе торможения. В процессе решения поставленных задач было установлено, что на форму переходного процесса влияет момент начала торможения и его длительность, начальный уровень давления в рабочей (поршневой) полости. Наиболее существенным результатом является определение структурных схем пневмопривода и алгоритмов их управления, позволяющих реализовать режим рекуперации энергии в магистраль в период торможения, что обеспечивает позитивный эффект энергосбережения, а также аналитическое описание источников экономии сжатого воздуха при работе привода. Преобразование кинетической энергии движущихся масс в потенциальную энергию сжатого воздуха осуществляется в штоковой полости пневмоцилиндра в момент торможения. Воздух, с давлением выше давления в магистрали, выходящий из поршневой полости, преодолевает силу сопротивления пружины обратного клапана, установленного в линии сброса сжатого воздуха, и поступает в магистраль. Значимость полученных результатов состоит в том, что полученные схемы с торможением за счет изменения структуры подключения полостей пневмоцилиндра, позволяют реализовать энергосберегающий режим работы, при котором кинетическая энергия движущихся масс привода переходит в потенциальную энергию сжатого воздуха и рекуперируется в сеть. Это дает возможность дальнейшего использования этой энергии, повышая уровень энергоэффективный режим работы.

Ключевые слова: пневматический привод, алгоритм управления, коммутация, рекуперация энергии, переходный процесс.

INTRODUCTION

Pneumatic drives are now widely used in industrial automation systems due to their practicality, especially before reaching speed limits, and the need to maintain productivity levels in production. At very high speeds, it becomes necessary to use additional methods to decelerate the cylinder piston at the end of the working stroke in order to prevent impacts against the cylinder cover and pneumatic rebound.

RESEARCH OVERVIEW

Traditional piston deceleration methods used in pneumatic drives lead to irreversible energy losses, which reduce the system's energy efficiency and increase production costs. The main methods for reducing impact forces are:

- Throttle regulation: by installing a throttle in the supply and exhaust lines, the airflow rate to and from the cylinder is regulated, allowing smoother piston movement and reducing sharp fluid movements [1]. Advantages of this method include hardware simplicity, low cost, and the ability to adjust flow rates. However, it does not provide complete shock absorption and is not energy efficient.
- Use of external shock absorbers: installed at the end positions of the piston stroke, external shock absorbers absorb the kinetic energy of the moving mechanical parts [2-4]. This method ensures high deceleration efficiency and allows work with significant loads, but it is also not energy efficient.
- Electro-pneumatic deceleration: combining pneumatics with electronic control systems enables dynamic adjustment of cylinder fluidity based on system parameters, allowing precise control of the deceleration process [5-7]. The advantages of this method include high deceleration accuracy and the ability to implement an energy-saving mode.

Analysis of literature sources shows that the moment of braking initiation and its duration significantly influence the dynamic characteristics of pneumatic and hydraulic systems. In particular, study [8] demonstrates that the parameters of the pneumatic accumulator, such as volume and initial pressure, determine the speed and stability of braking, thereby affecting the shape of the transient process.

An analysis of actuator performance under conditions of variable physical parameters of the working medium allows for improvements in control and braking methods. Several studies [9, 10] have examined factors that alter system dynamics, such as the presence of gas inclusions and the specific motion characteristics of the working body. The results of these studies can be used to develop braking strategies for pneumatic drives by adaptively adjusting switching configurations to ensure smooth and stable deceleration. Study [14] investigates pressure delay in pneumatic braking systems caused by the influence of pipelines. It was found that pipeline length and diameter can lead to significant delays in pressure transmission, which in turn impacts braking accuracy and efficiency. Research [15] focuses on the use of predictive control models that enable precise pressure regulation in the brake chamber, ensuring a stable transient process even under varying braking conditions.

For instance, initiating braking too early or over an extended distance can significantly degrade the quality of the transient process [16-18].

A key challenge is to mathematically determine the optimal moment to initiate braking and its appropriate duration (i.e., a specific combination of switching scenarios for cylinder chambers) for each particular drive configuration [19-22]. Studies [23-25] show that the use of fast-acting bistable valves, combined with a switching control algorithm, can substantially improve braking performance by reducing system response time and enhancing the transient process profile.

Therefore, it is necessary to analytically justify and carry out computer modeling of the operating processes within the pneumatic system.

The aim of the study is to assess the influence of the pneumatic system structure and the choice of control algorithm on the feasibility of implementing an energy-saving operating mode.

RESEARCH RESULTS AND THEIR DISCUSSION

The main challenge in computer modeling of the transient process in pneumatic drives with braking achieved by altering the structure of switching connections is determining the optimal switching moment of the braking pneumatic distributor. This moment must ensure shock-free drive operation and eliminate pneumatic rebound of the piston.

The method for calculating the optimal braking distance is based on numerical integration of the system of differential equations (1) using the Runge-Kutta step method. As a result of integration at the *i*-th step, the following values are known: p_{1i} and p_{2i} - pressure in the working and exhaust chambers, x_i - piston position, and v_i - piston velocity. After obtaining these values, the computational algorithm determines the optimal position of the brake distributor. At each integration step, the algorithm predicts the ratio of the gas expansion work and the kinetic energy of the moving parts to the work of the forces opposing the piston's motion, assuming that the brake distributor is activated at that moment.

$$\begin{cases} \dfrac{dp_1}{dt} = \alpha \dfrac{k\, f_1^e \beta p_s \sqrt{RT_s}\,\varphi\left(\dfrac{p_1}{p_s}\right)}{F_1\left(x_{01}+x\right)} - \beta \dfrac{k\, p_1}{x_{01}+x}\cdot\dfrac{dx}{dt}; \\ \dfrac{dp_2}{dt} = -\gamma \dfrac{k\,\beta f_2^e \sqrt{RT_s}\, p_2^{\frac{3k-1}{2k}}\varphi\left(\dfrac{p_a}{p_2}\right)}{F_1\left(x_{01}+x\right)} + \\ + \delta \dfrac{k\, p_2}{L+x_{02}-x}\cdot\dfrac{dx}{dt}; \\ \dfrac{dx}{dt} = v; \\ \dfrac{dv}{dt} = \dfrac{1}{m}\left(p_1F_1 - p_2F_2 - P\right), \end{cases} \tag{1}$$

where m – the mass of moving parts of the drive; P – the static load on the rod; F_1, F_2 – the piston area on the side of the rodless and rod cavities; f_1^e, f_2^e – effective areas of the intake and exhaust lines; k – the adiabatic index; R – the gas constant; T_s – the temperature in the supply line; $\varphi(...)$ – the cost function; L – the full stroke of the piston; x_{01}, x_{02} – initial coordinates of the piston on the left and right; p_s – the pressure in the supply line; p_a – the atmospheric pressure; α , β , γ , δ – tags that take the value 1 or 0 depending on the phase of the drive movement.

When switching the brake distributor $\gamma = \alpha = 0$ in the system of equations (1) and integrating the first two remaining equations of system (1), it leads, respectively, to the adiabatic expansion equation and the adiabatic compression equation (2).

$$\begin{cases} p_1\left(x_{01}+x\right)^k = const; \\ p_2\left(L+x_{02}-x\right)^k = const. \end{cases} \qquad (2)$$

Integrating the last equation of system (1) taking into account (2) allows us to obtain all components of the energy balance of the braking process (3).

$$m\int_{v_i}^{v_L} d\left(\frac{v^2}{2}\right) = \int_{x_i}^{L} F_1 p_1 dx - \int_{x_i}^{L} F_2 p_2 dx - \int_{x_i}^{L} P dx. \qquad (11)$$

The values of individual components of expression (3) are denoted by T_i, Π_i, Π_{1i}, A_i (4).

$$\begin{cases} \Pi_i = \int_{x_i}^{l_i+x_i} F_1 p_1 dx + p_k F_1 \int_{l_1+x_i}^{L} dx = \frac{F_1 p_{1i}\left(x_{01}+x\right)}{k-1}\times \\ \times\left[1-\left(\frac{x_{01}+x_i}{x_{01}+x+l_1}\right)^{k-1}\right] + p_k F_1\left(L-x-l_1\right); \\ \Pi_{1i} = \int_{x_i}^{L-l} F_2 p_2 dx + p_s F_2 \int_{L-x}^{L} dx = \\ = \frac{F_2 p_{2i}\left(L+x_{02}-x_i\right)}{k-1}\times \\ \times\left[\left(\frac{L+x_{02}-x_i}{x_{02}+l}\right)^{k-1}-1\right] + F_2 p_s l; \\ A_i = \int_{x_i}^{l} P dx = P\left(L-x_i\right); \\ T_i = \frac{mV_i^2}{2}, \end{cases} \qquad (4)$$

$$l = \left(\frac{p_{2i}}{p_s}\right)^{1/k}\left(L+x_{02}-x_i\right) - x_{02}; \qquad (5)$$

$$l_1 = \left(\frac{p_{1i}}{p_s}\right)^{1/k}\left(x_{01}+x_i\right) - \left(x_{01}+x_i\right), \qquad (6)$$

when p_k – the pressure reducing valve setting.

In the system (4): A_i – the work required to overcome static resistance forces; T_i – the kinetic energy of the moving parts; Π_i – the potential energy of the expansion of compressed air in the working chamber of the cylinder along the braking path, taking into account the activation of the two-line pressure-reducing valve; Π_{1i} – the potential energy of the compression of air in the exhaust chamber of the cylinder along the braking path, considering the activation of the check valve; l – the distance, which traveled by the piston with the check valve open; l_1 – the distance, which traveled by the piston from the beginning of braking until the activation of the pressure-reducing valve.

The operation of the drive with the braking system shown in Fig. 17 is modeled as follows: at the beginning of the program some control symbols K and W values are given $K=0$, $W=1$.

If $W=1$, then $\alpha=\beta=\gamma=\delta=1$ in the equation (1):

$$\begin{cases} \text{If } T_i+\Pi_i \ge \Pi_{1i}+A_i \wedge K=0 \text{ then } W=0, K=1; \\ \text{If } W=0 \wedge p_{1i} > p_k \text{ then } \alpha=0,\ \beta=1; \\ \text{If } W=0 \wedge p_{1i} \le p_k \text{ then } \alpha=0,\ \beta=0,\ p_{1i}=p_k; \\ \text{If } W=0 \wedge p_{2i} < p_s \text{ then } \gamma=0,\ \delta=1; \\ \text{If } W=0 \wedge p_{2i} \ge p_s \text{ then } \gamma=0, \delta=0,\ p_{2i}=p_s. \end{cases}$$

This technique allows the determination of the optimal braking distance and the complete transient process in a single calculation, under the condition of braking the piston movement by changing the switching states.

Let us carry out an analytical analysis of the braking process based on a scheme that enables the implementation of an energy-saving mode (Fig. 14), identify the sources of energy savings, and derive the dependencies that allow for the quantitative assessment of compressed air savings.

Based on equations (1–6), the optimal coordinate for the start of braking can be defined as the point at which the following equality is satisfied:

$$\Pi_i + T_i = \Pi_{1i} + A_i. \qquad (7)$$

In the braking phase, we can distinguish three sources of compressed air savings:

1. When the working chamber is blocked by the brake distributor, the movement of the working element is driven by the potential energy of the compressed air that entered the working chamber before braking. The distance l_1, traveled by the piston until the moment the

pressure-reducing valve, is activated is determined by expression (6).

Mass of air saved during this braking phase

$$M_1 = \frac{l_1 \cdot F_1 (p_s - p_a)}{R\,T_s}. \qquad (8)$$

2. After the pressure reducing valve is activated, the pressure level in the working cavity is maintained lower than the main pressure, thereby saving compressed air. The path that the piston travels in this braking phase

$$l_2 = L - (x_i + l_1) = L - \left(\frac{p_{1i}}{p_k}\right)^{1/k} (x_{01} + x_i) + x_{01}; \quad (9)$$

Mass of air saved during this braking phase

$$M_2 = \frac{l_2 \cdot F_1 (p_s - p_k)}{R\,T_s}. \qquad (10)$$

3. Return of compressed air from the exhaust (braking) cavity to the main after opening the check valve (recuperation).

The path traveled by the piston in the recuperation mode is determined by expression (5). The mass amount of air saved during this braking phase

$$M_3 = \frac{l \cdot F_2 (p_s - p_a)}{R\,T_s}. \qquad (11)$$

The combination of these three sources of compressed air savings results in significant overall savings when the pneumatic drive operates with an optimal structure and control algorithm. Moreover, the energy-saving effect becomes more pronounced as the braking distance increases, that is, with an increase in the mass load.

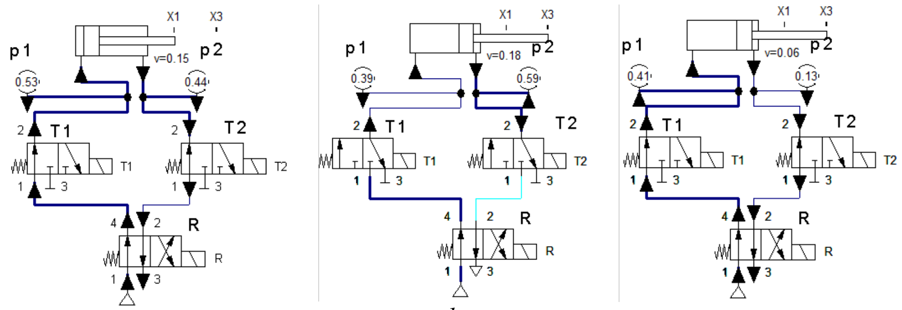
RESEARCH METHODS AND EXAMPLES

Through analysis, we identified the most promising pneumatic system configurations and examined their switching mechanisms (Figs. 1, 3, 5, 7 and 14). Subsequently, we modeled the operation of pneumatic actuators in the FluidSim software environment, generating graphs depicting movement, velocity, acceleration, and pressure variations within the working (p_1) and exhaust (p_2) cavities (Figs. 2, 4, 6, 8 and 15). The configurations are arranged in order of increasing energy efficiency. The model parameters for all configurations were consistent: the piston diameter $d_\text{p} = 100$ mm; the rod diameter $d_\text{r} = 40$ mm; the piston stroke $L = 400$ mm; moving mass $m = 350$ kg (referenced to the piston's axis of inertia); the piston load $P = 700$ N; the supply line pressure $p_\text{s} = 0{,}6$ MPa.

Scheme *A* (fig. 1) is controlled by three 3/2 pneumatic distributors. During acceleration, the piston cavity is connected to the supply line, as in scheme *A*, and the exhaust cavity – to the atmosphere. When moving from left to right during braking, only distributors T_1 and T_2 are activated (diagram *b* in Fig. 1), completely blocking both the air supply and exhaust from the cylinder chambers.

Since the actuator has a significant moving mass (350 kg), which creates a strong inertial effect, the piston continues to move to the right during braking (Fig. 2). As a result, the air in the brake chamber is compressed, the pressure increases rapidly (reaching values more than three times the main pressure), which leads to a sharp deceleration of the piston.

Commutation situation	T_1	T_2
Acceleration	0	0
Braking	1	1
Positioning	0	0



a　*b*　*c*

a – acceleration, b – braking, c – positioning

Fig. 1. Pneumatic actuator structure, distributor actuation map and cylinder cavity switching diagrams (scheme *A*).

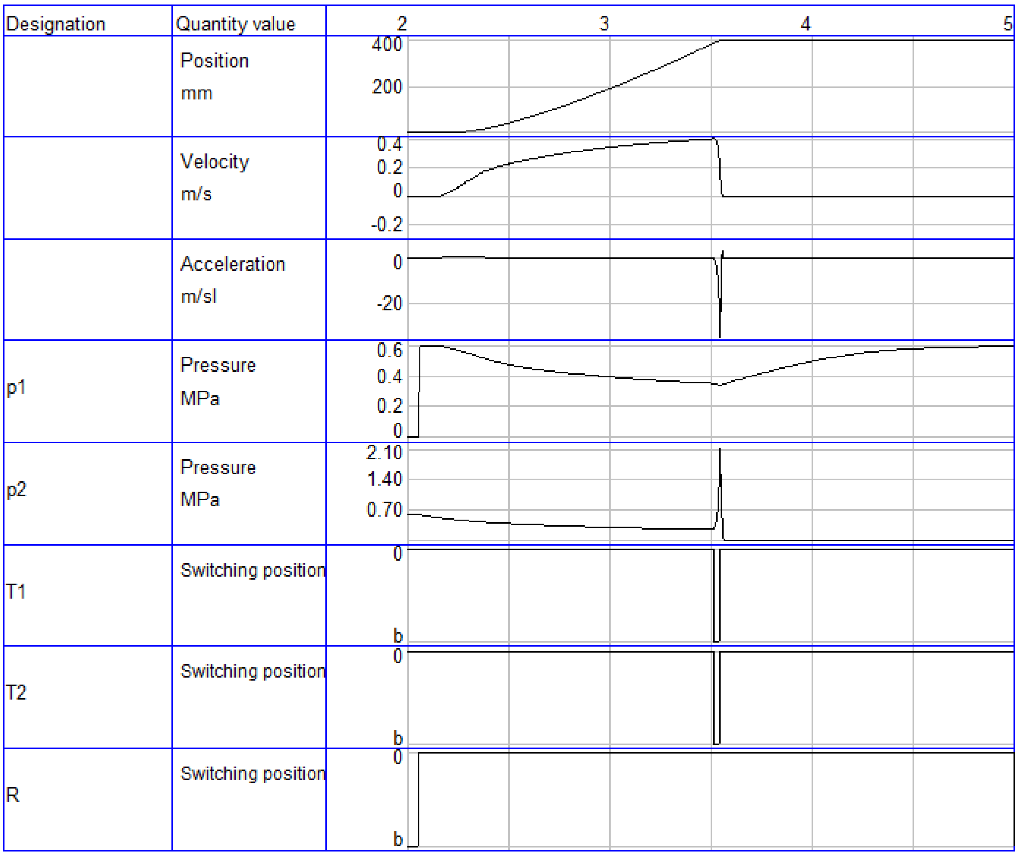


Fig. 2. Oscillogram of the transient process in a pneumatic actuator (scheme *A*, flow rate through the distributors *R* = 1000 l/min, T_1 = 550 l/min, T_2 = 850 l/min).

To prevent pneumatic rebound at the moment of piston fixation (diagram *c* in Fig. 1), the working chamber is reconnected to the supply line, while the brake chamber is connected to the atmosphere.

The main disadvantage of this scheme is the sudden drop in piston speed. However, its advantage is the short duration of the transition process – approximately 1.5 seconds.

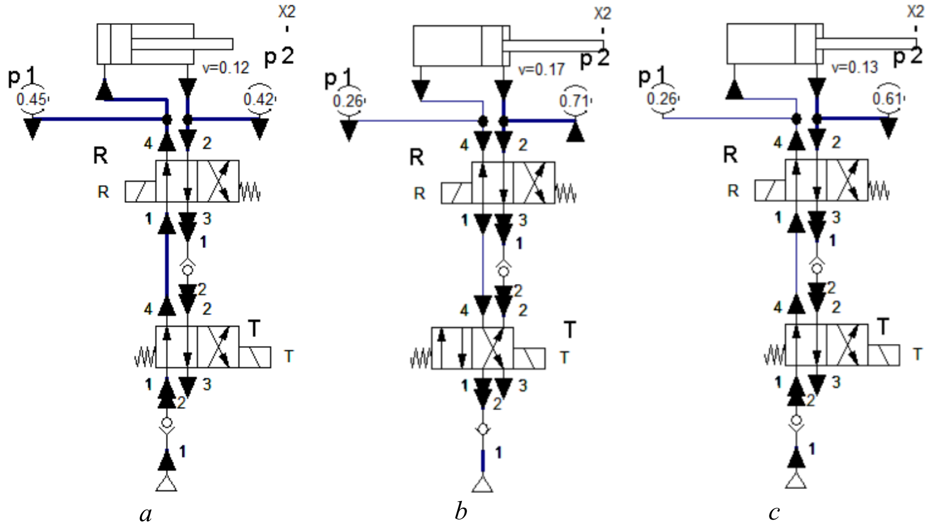
In summary, we can conclude that scheme *A* is applicable, as they achieve a rational form of the transient process even in the presence of large moving masses (350 kg). However, the second objective of the research – saving compressed air – cannot be achieved using these scheme.

Fig. 3 shows scheme *B*, which is implemented on the basis of two 4/2 pneumatic distributors and contains check valves in the supply line and exhaust line, due to which it is possible to achieve the effect of energy saving: in the final period of piston movement, compressed air from the exhaust line (where at a certain time interval due to significant inertia forces of the drive the pressure exceeds the main line) enters the network. This effect is achieved due to the switching of the rod cavity during the braking period with the supply line (Fig. 3, *b*), when the excess pressure (level higher than 0.6 MPa) overcomes the resistance of the check valve and returns to the network, and the piston cavity at this time is connected to the atmosphere.

The advantages of scheme *B* also include maintaining a constant minimum pressure drop across the piston throughout the entire acceleration phase (approximately two-thirds of the stroke) and ensuring a smooth decrease in piston speed during the positioning phase, without pneumatic rebound (Fig. 4).

Scheme *C* (Fig. 5) is based on the 4/2 distributor *R* and the 5/2 distributor *T*. The key difference in this scheme is the presence of a pressure-reducing valve in the supply line, which is active only during the braking phase (Fig. 5, *b*).

Commutation situation	*R*	*T*
Acceleration	1	0
Braking	1	1
Positioning	1	0



a – acceleration, b – braking, c – positioning

Fig. 3. Pneumatic actuator structure, distributor actuation map and cylinder cavity switching diagrams (scheme *B*).

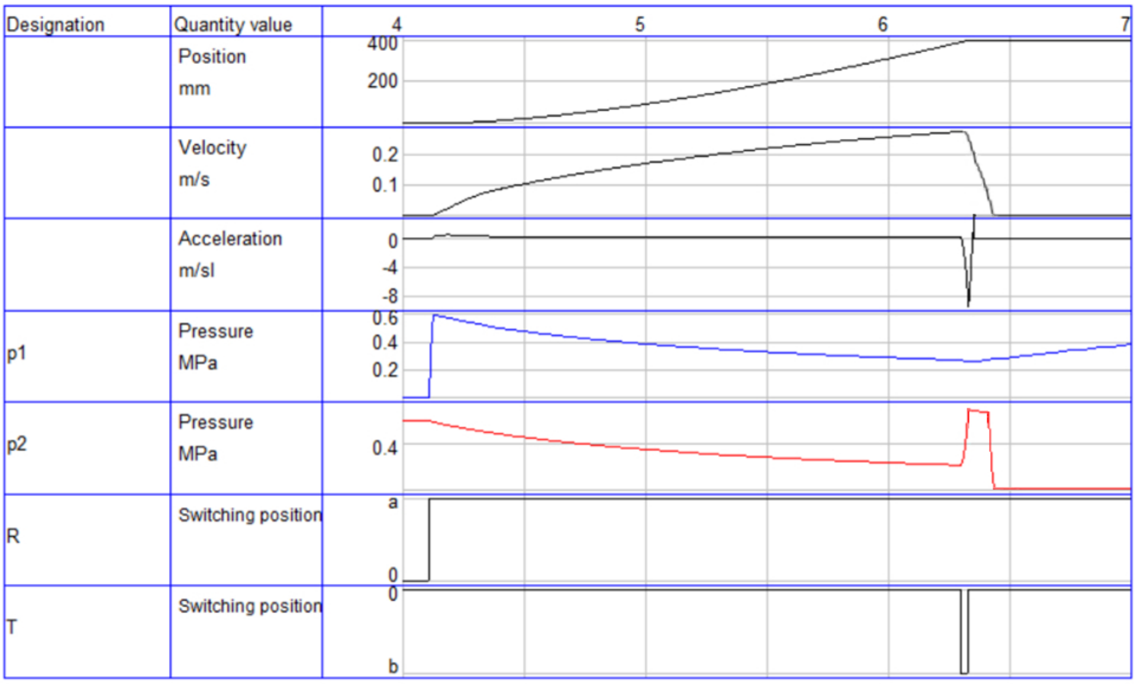


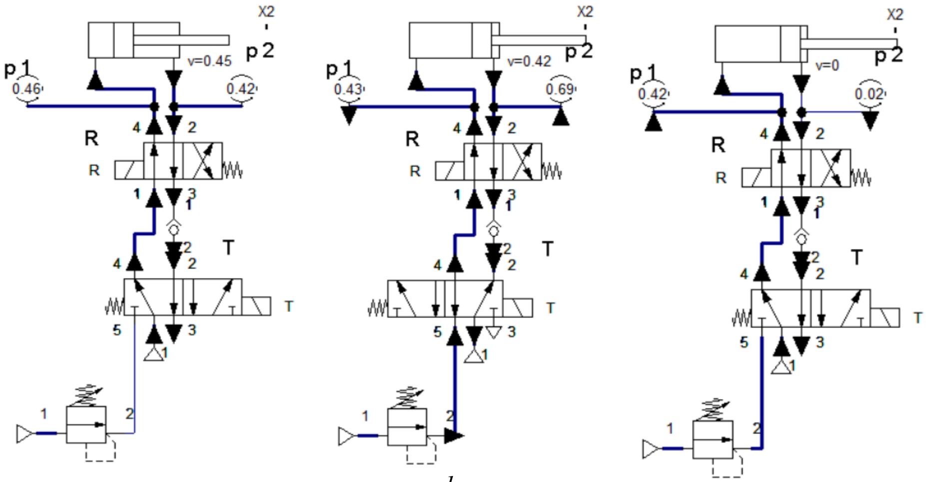
Fig. 4. Oscillogram of the transient process in a pneumatic actuator (scheme *B*, flow rate through the distributors *R* = 600 l/min, *T* = 800 l/min).

In this case, it is possible to maintain the pressure in the working cavity at a level equal to the setting of the pressure-reducing valve, while the excess pressure from the brake cavity flows back to the main line (overcoming the resistance of the check valve), thereby achieving an energy-saving effect (Fig. 6).

The simplest scheme in terms of hardware implementation is Scheme *D* (Fig. 7), which includes only a single 4/2 pneumatic distributor, a power supply, and a cylinder.

However, despite its simplicity, this design enables a transient process that is quite efficient in terms of a smooth decrease in piston speed (Fig. 8) and, by directing excess pressure back to the line during braking, achieves an energy-saving effect. Another advantage of this pneumatic actuator design is that the transient duration is approximately one second, whereas in previous schemes, it ranged from 1.5 to 2 seconds.

/	R	T
Acceleration	1	0
Braking	1	1
Positioning	1	0



a *b* *c*

a – acceleration, b – braking, c – positioning

Fig. 5. Pneumatic actuator structure, distributor actuation map and cylinder cavity switching diagrams (scheme *C*).

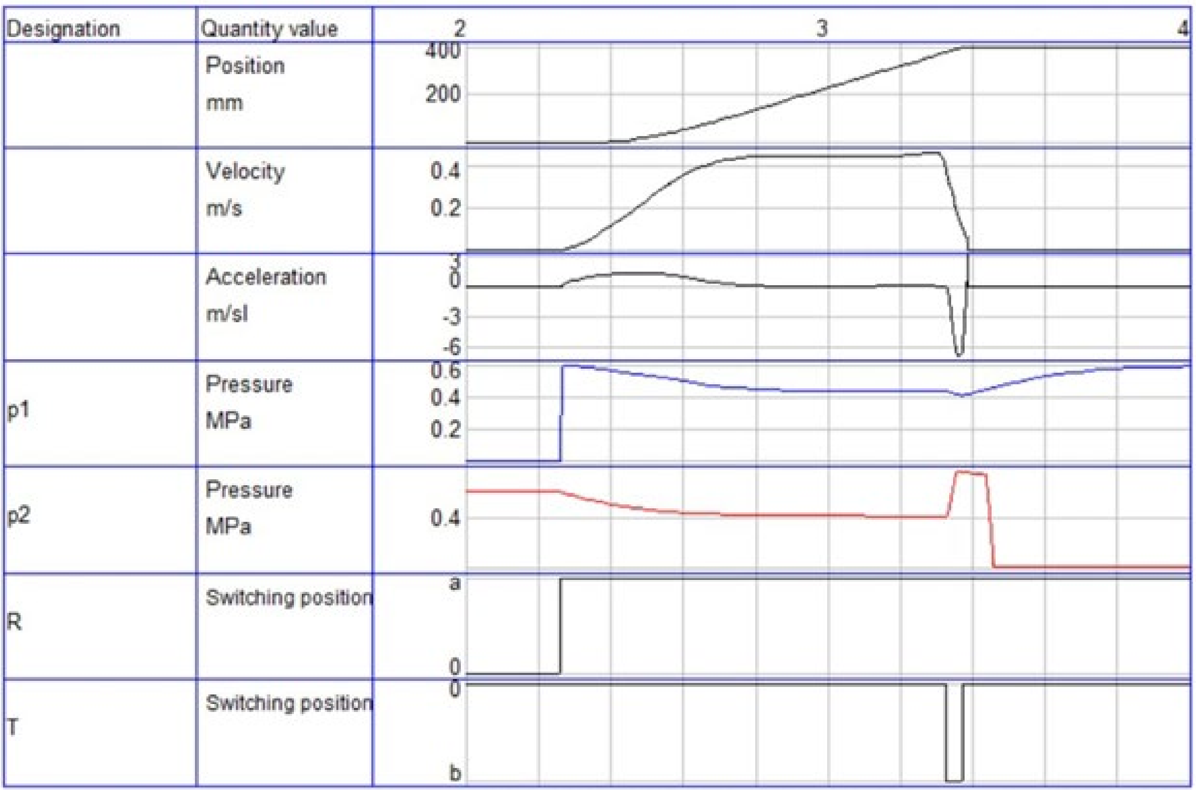
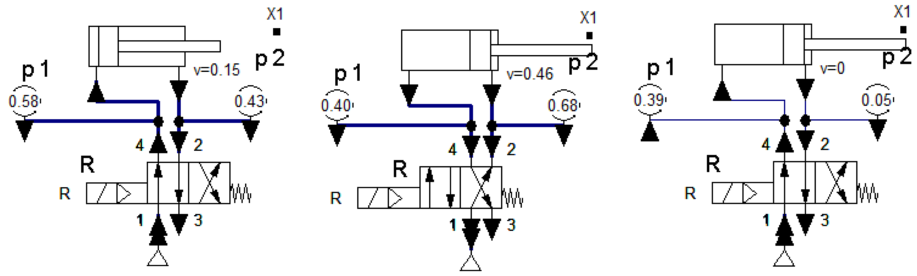


Fig. 6. Oscillogram of the transient process in a pneumatic actuator (scheme *C*, flow rate through the distributors *R* = 1200 l/min, *T* = 1200 l/min, *L* = 370-385 mm).

The main drawback of scheme *D* is that during the braking phase, the working cavity is connected to the atmosphere, causing a pressure drop. Then, during the locking phase, when the cavity reconnects to the supply line, the pressure rises again. This results in additional air consumption during operation and reduces the overall energy efficiency of the system.

The proposed methods of braking and positioning the piston offer a wide range of schemes and options for switching the working and exhaust chambers of the pneumatic cylinder

Commutation situation	*R*	*T*
Acceleration	1	-
Braking	0	-
Positioning	1	-



a *b* *c*

a – acceleration, b – braking, c – positioning

Fig. 7. Pneumatic actuator structure, distributor actuation map and cylinder cavity switching diagrams (scheme *D*).

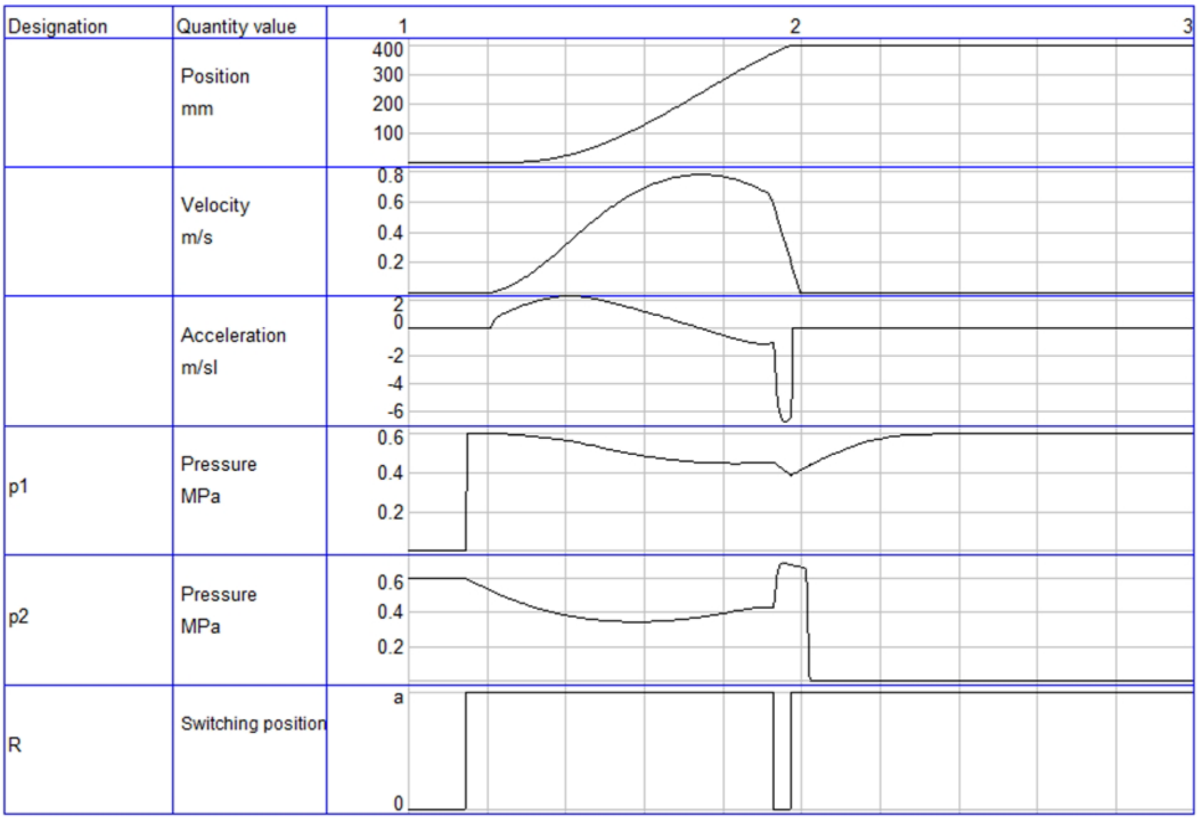


Fig. 8. Oscillogram of the transient process in a pneumatic actuator (scheme *D*, flow rate through the distributor *R* = 1500 l/min, *L* = 355-390 mm).

during acceleration, braking, and fixation phases of the working element.

It is necessary to determine the most rational circuit solution for a pneumatic drive that utilizes braking by changing the structure of switching connections. Based on the principle of energy conservation, the scheme and the control algorithm must satisfy the following conditions:

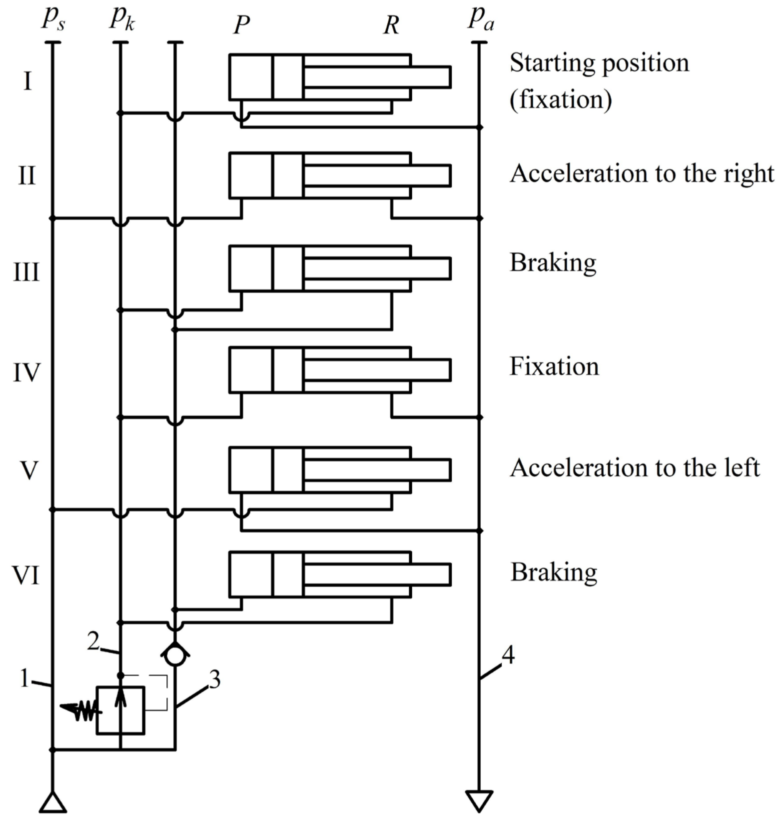
– In the fixation phase, the piston must be held by the minimum required pressure differential across the piston, consistent with the counter load. In this case, the lower pressure must correspond to atmospheric pressure, while the higher pressure should be supplied from the outlet of the pressure-reducing valve. Moreover, this pressure must be significantly lower than the main line pressure.

– In the acceleration phase, the working chamber, previously connected to the atmosphere, is switched to the supply line (pressure p_s), and the exhaust chamber, previously connected to a source of reduced pressure p_k, is connected to the atmosphere. The low backpressure in the exhaust chamber ensures rapid piston acceleration. In this case, the unproductive work of pushing compressed air from the exhaust chamber is minimized.

– In the braking phase, the working chamber must be connected to the reduced pressure source p_k (to the outlet of the two-line pressure-reducing valve), while the exhaust chamber, through a previously closed check valve, is connected to the main supply line with pressure p_s. In the working chamber, the compressed air expands to the pressure p_k, and once the pressure in the brake chamber increases to the main pressure p_s, the air from the brake chamber returns to the network.

Fig. 9 shows an expanded diagram corresponding to the most rational switching connections.

Let us imagine all possible switching situations in the form of a connection graph (Fig. 10), where the vertices correspond to each of the six possible switching situations of the drive (P – piston cavity of the pneumatic cylinder, R – rod cavity).



a – acceleration, b – braking, c – positioning

Fig. 9. Exploded diagram of the energy-saving structure of the pneumatic system.

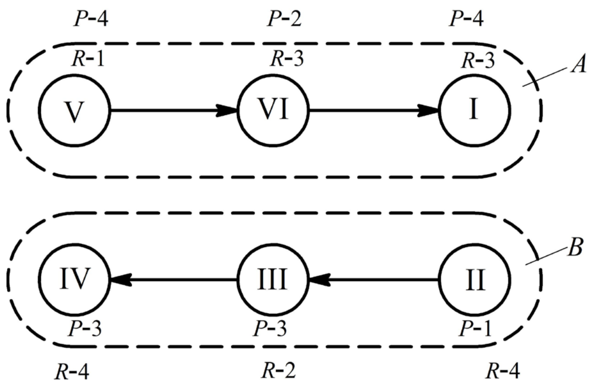


Fig. 10. Graph of switching situations.

The situation graph (Fig. 10) is divided into macrosituations *A* and *B*, corresponding to the extension and retraction of the rod, respectively. Situations V, VI, and I(A) share a common characteristic – rod retraction, while situations IV, III, and II(B) correspond to rod extension. Therefore, it is quite evident that at least a four-line, two-position distributor must be used, connected to the cylinder chambers (Fig. 11). Each of the situations necessarily occurs within both macrosituations. Consequently, each of the two inlets of the four-line distributor can be connected to the outlet of a three-line distributor, as illustrated in Fig. 12.

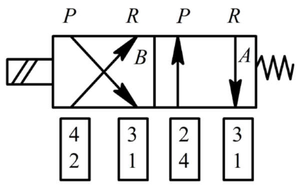
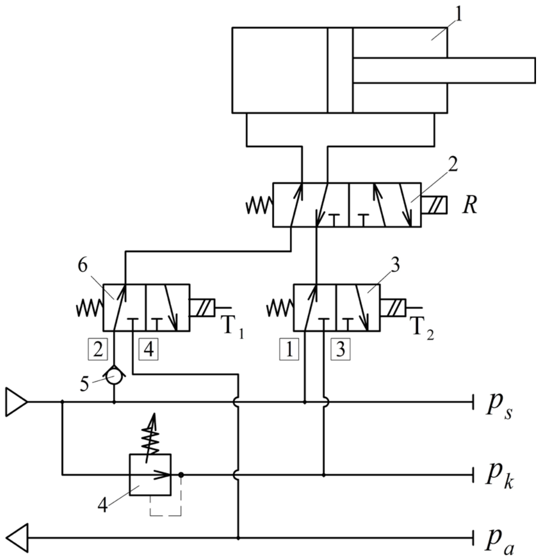


Fig. 11. Optimal cylinder connection option



	Movement phases	T_1	T_2	R
Move to the right	Initial state	1	1	0
	Acceleration	1	0	1
	Braking	0	1	1
	Positioning	1	1	1
Move to the left	Initial state	1	1	1
	Acceleration	1	0	0
	Braking	0	1	0
	Positioning	1	1	0

Fig. 12. Energy-saving pneumatic actuator scheme based on a 5/2 valve

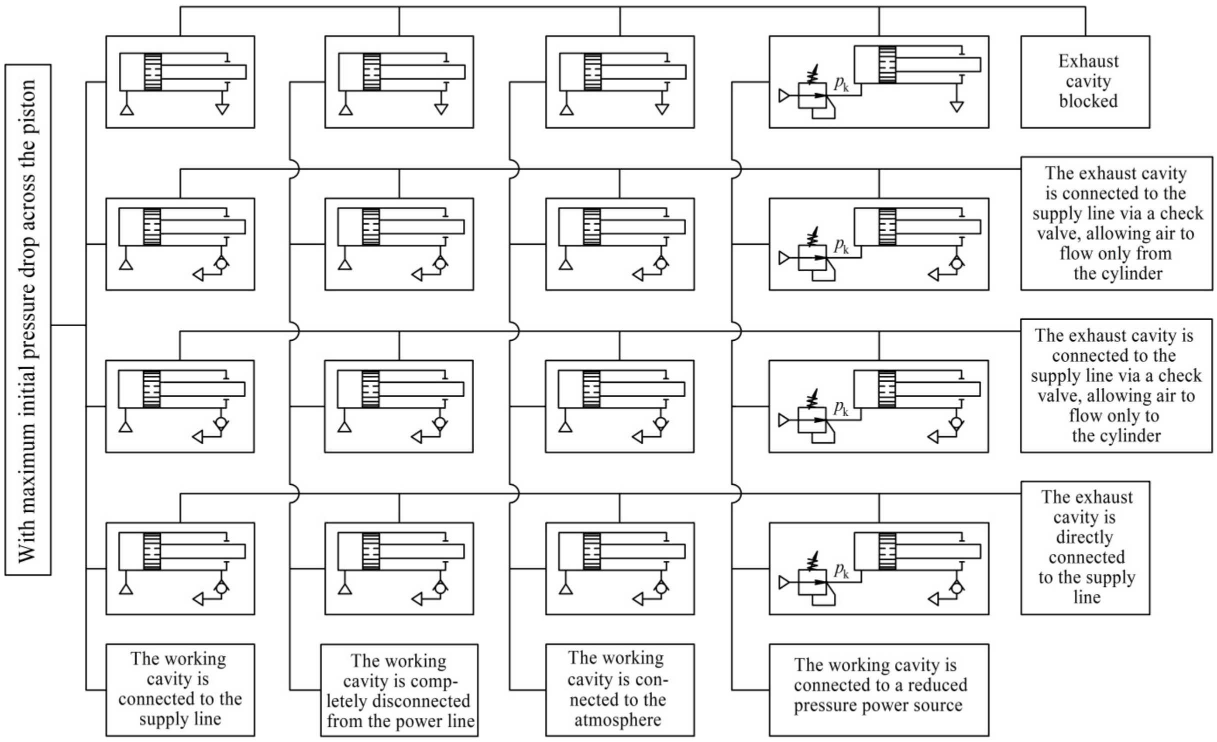
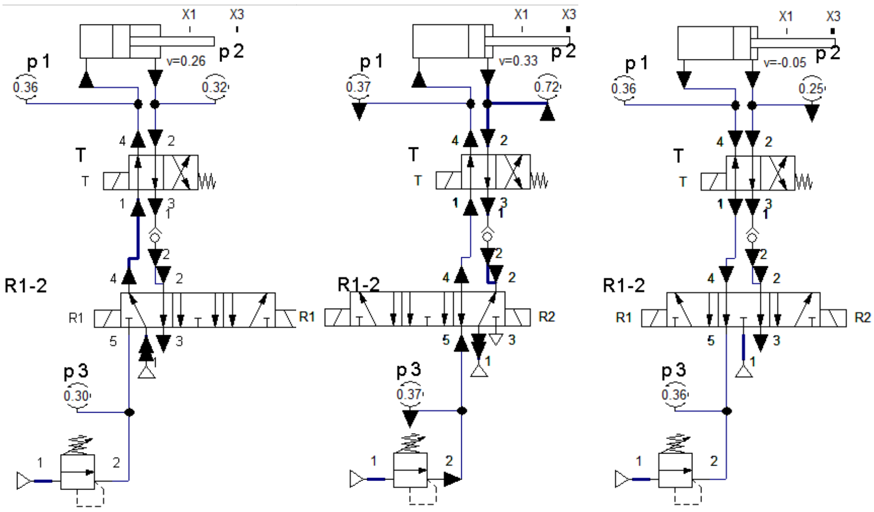


Fig. 13. Table of possible switching situations of the pneumatic actuator.

Commutation situation	T	R_1	R_2
Acceleration	1	1	0
Braking	1	0	1
Positioning	1	0	0



a *b* *c*

a – acceleration, b – braking, c – positioning

Fig. 14. Pneumatic actuator structure, distributor actuation map and cylinder cavity switching diagrams (scheme *E*).

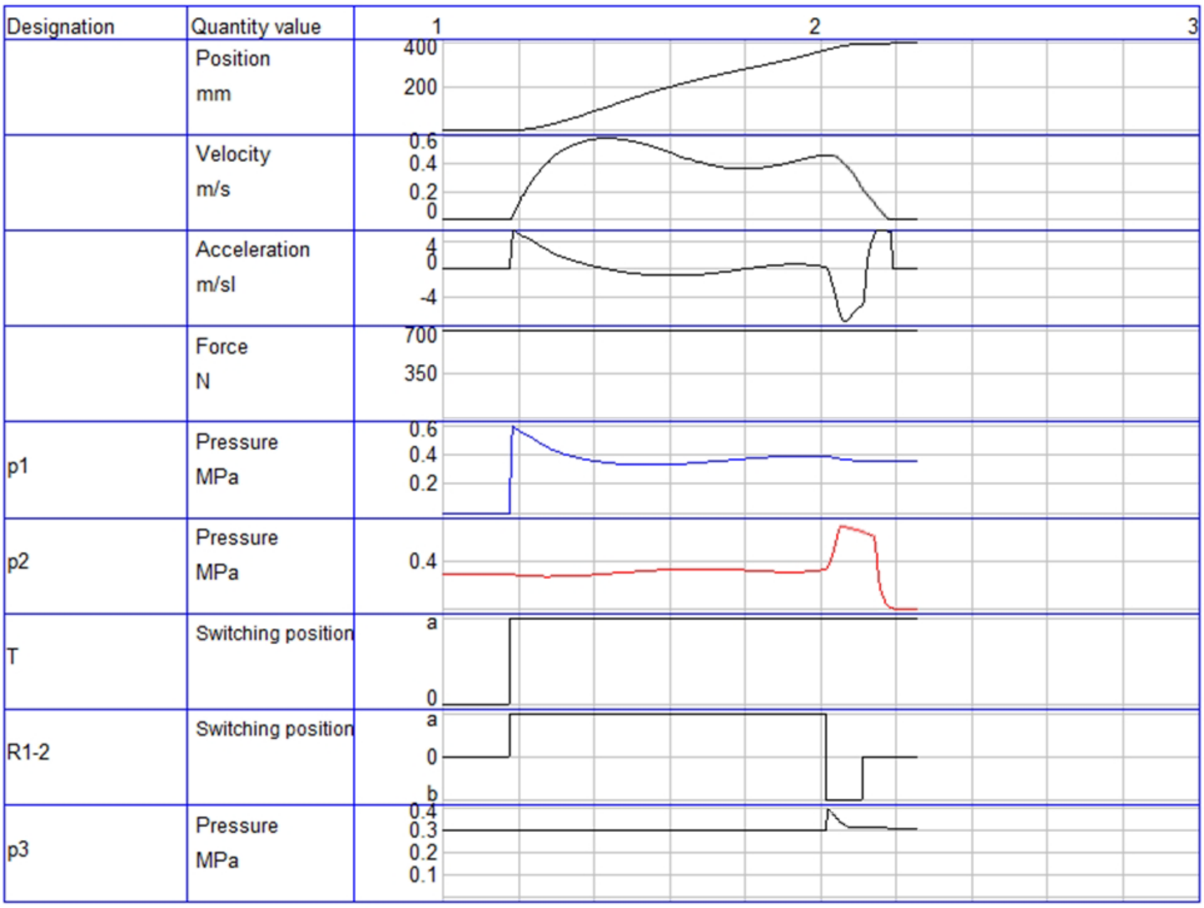


Fig. 15. Oscillogram of the transient process in a pneumatic actuator (scheme *E*, flow rate through the distributor T = 1000 l/min, R = 1000 l/min, L = 360-385 mm).

Based on the above general data from the conducted research, a table of possible switching situations of the pneumatic system was formed (Fig. 13). According to Fig. 13, a drive diagram was obtained that allows implementing the energy saving mode (energy recovery to the network) (Fig. 14).

Research data (Fig. 15) show that scheme *E* (Fig. 14), built on the basis of the above conclusions, is one of the most effective in terms of achieving the goal of a short duration of the transient process (high piston speed), a smooth process of its braking and the implementation of the recuperation mode.

This effect is achieved both by a successful selection of the hardware and by adjusting the pressure of the pressure reducing valve, which allows at the initial moment of movement to obtain a pressure in the piston chamber equal to the pressure setting of the pressure reducing valve (0.3 MPa), which has a positive effect on the transient process: the speed is more stable, during the acceleration period (approximately 60% of the path) the pressure drop across the piston is close to constant (Fig. 15).

During braking (Fig. 14, b), the working cavity is connected to the outlet of the pressure reducing valve, which maintains a stable pressure level in it, and the exhaust cavity is connected to the main, where an excess amount of air enters, the pressure of which exceeds 0.6 MPa.

Based on the analysis of the transient process in Fig. 15, the following conclusions can be drawn regarding the advantages of scheme *E*:

- on the graph of pressure changes in the exhaust cavity, there is a section in which the recovery of excess pressure into the network occurs;
- the scheme allows for the implementation of a uniformly slowed and regulated braking mode (the magnitude of the negative acceleration is regulated by adjusting the pressure of the pressure reducing valve).

Therefore, scheme *E* allows for the full achievement of the research goal: to obtain the optimal form of the transient process both in terms of speed and smoothness of piston braking and, at the same time, to obtain an energy saving effect due to the recovery of compressed air into the network during the braking period. Quantitatively estimate the mass fraction of saved compressed air can be calculated by formulas (9, 11, 12) depending on the specific parameters of the drive and the duration of the transient process. For the scheme with the calculated parameters shown in Fig. 14-15, the savings are about 15%.

The presented schemes with braking due to a change in the structure of switching connections have a fundamental difference from schemes with a traditional structure (with throttle braking): during throttle braking, the kinetic energy of the moving parts of the drive is converted into thermal energy and is irreversibly lost, and in the proposed schemes, the kinetic energy of motion is converted into the potential energy of compressed air. This provides significant opportunities for the further use of this energy.

CONCLUSIONS

As a result of modeling the operation of a pneumatic actuator with different braking schemes in the «Fluidsim» software environment, the following findings were made:

1. During braking, the shape of the transient process in a pneumatic actuator is significantly influenced by the braking path control algorithm. For example, if the braking process is initiated too early in scheme *B* or carried out over an extended distance, it considerably degrades the quality of the transient response.
2. The initial pressure level in the working (piston) chamber also has a substantial impact on the transient process quality. When the initial pressure is reduced-achieved through the use of a pressure-reducing valve in the supply line - the piston speed and pressure change more smoothly compared to cases with full initial supply pressure. The most favorable conditions are observed when the working chamber pressure is regulated via a pressure-reducing valve, as seen in schemes *B* and *E*.
3. The kinetic energy of the moving parts of the actuator is converted into the potential energy of compressed air and, in energy recovery mode, is returned to the main line during the braking phase. This is made possible through the use of pneumatic actuator configurations that incorporate a pressure-reducing valve in the supply line, a check valve in the exhaust line, and appropriate control algorithms.

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